PATENT SPECIFICATION (11) 1 593 852

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(21) Application No. 46252/76 (22) Filed 5 Nov. 1976 (19)

(23) Complete Specification filed 4 Nov. 1977

(44) Complete Specification published 22 July 1981

(51) INT. CL.³ F02M 31/02 25/00

(52) Index at acceptance

F1B 2F18A 2F7 2L1A 2L3A2 2L3B2 B106 B120 B145 B200 B212 B214 B226 B300 B3!0 BH

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(54) IMPROVEMENTS IN OR RELATING TO COMPRESSION-IGNITION ENGINES

(71) We, BURMAH OIL TRADING LIMITED, a British Company of Burmah House, Pipers Way, Swindon, Wiltshire SN3 1RE, do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

This invention relates to compression-ignition engines and to methods of operating the same.

The world possesses finite stocks of crude oil and thus efforts are being made to find alternative fuels and sources of fuels and to make more efficient use of the dwindling stocks of crude oil and natural gas. Processes are available for converting natural gas, much of which is now burnt at the oilfield, into methanol which is thus potentially available in large quantities as a fuel at a moderate cost. There are also processes for making methanol from coal and thus methanol may be a moderately priced fuel in areas where there are large enexploited deposits of coal.

Although methanol has been used in special circumstances (e.g. "drag car" racing) as a fuel for a spark-ignition engine, either as the sole fuel or as an addition to petrol, it has generally been considered to be unsuitable for use in a compression-ignition engine since methanol/air mixtures will not self-ignite at the sort of temperatures and pressures that are usually achieved during the compression stroke of such an engine. Typically the temperature of the air compressed during the compression stroke of a compression-ignition engine may reach as high as 500°C, a temperature at which under the pressure thus prevailing in the cylinder a diesel fuel/air mixture will self-ignite. On the other hand petrol/air mixtures may require a temperature of about 1000°C before they will self-ignite at such pressures, and methanol/air mixtures will also need a much higher temperature than 500°C for self-ignition upon compression.

Increasingly demands are being made that pollution of the environment by exhaust emission from road vehicles must be reduced. There is a considerable amount of research going on in a number of countries directed towards developing fuels and engines that will give rise to reduced exhaust emission, particularly as regards reduced emission of smoke, lead compounds (from the lead-containing anti-knock additives), carbon monoxide, nitrogen oxides and unburnt fuel.

There is also increasing pressure on the manfuacturers of diesel engines to produce units that will produce more power for a given weight or volume of engine. One approach that has been tried is to fit to the air intake a turbo-charger which is driven by the exhaust gases. In this way a greater weight of air is drawn into the cylinder and hence more fuel can be burnt during each cycle, thus increasing the work output of the engine. The incoming air is heated by compression in the turbo-charger and its temperature is raised still further by heat transfer from the exhaust gases used to drive the turbo-charger. It is therefore usual to fit an intercooler in the path of the air from the turbo-charger. There have also been proposals to fit two turbo-chargers in series in order to obtain increasee power from a diesel engine each with its own intercooler. Intercoolers are bulky and may give rise to design problems in fitting a diesel power unit in a confined space, such as in a diesel-powered railway locomotive.

A report has appeared (E. Mühlberg, Automobilstechnisch Zeitschrift, January, 1963, Vol. 65, No. 1, P16—24) describing some preliminary tests in whch a diesel engine was operated using the so-called "Carburettor-Diesel" method. In the "Carburettor-Diesel" method alcohol is introduced into the combustion chamber by carburetting the alcohol into the induced air and ignition is effected in the usual way be injected diesel oil or other easily ignited fuel. As shown in Figure 2 of that paper, knock-free combustion was achieved with at most 66% by weight of methanol (i.e. 50% on an energy basis) as a

percentage of the whole fuel quantity (diesel fuel and methanol) under certain specified test conditions. The author of the paper then goes on to discuss the use of partial exhaust gas return as a means to enable the MAN—M model multifuel engine to run satisfactorily on alcohol alone without any additive, either more easily ignited fuel constituent such as diesel fuel or an ignition promotor.

It has now been found that a diesel engine can be run satisfactorily, particularly as regards smoke emission characteristics using the two fuels methanol and a diesel fuel and with methanol supplying more than 66% by weight of the total fuel quantity (corresponding to more than 50% on an energy bais), if the air or the methanol air mixture is preheated using waste heat from the engine.

According to the present invention in one aspect a method of operating a compression-ignition engine having at least one combustion space comprises the steps of providing a mixture comprising methanol and air in the said combustion space, compressing the mixture and injecting a quantity of a diesel fuel into the at least partially compressed mixture, the temperature of the air being raised by at least 20°C prior to compression, by preheating with waste heat from the engine.

Preferably the temperature is increased by 50 to 70°C, or higher e.g. 90°C.

When the methanol is introduced into the air inlet path the air supplied to the combustion space is cooled as a result of the high latent heat of evaporation of methanol. Indeed the cooling can be so great as to cause deposition of ice on the air intake passage arising from condensation of the moisture in the air. If the engine is fitted with a turbo-charger it may be possible to reduce the size of the intercooler or to dispense with it altogether under favourable circumstances. However the air cooling effect has the disadvantage that the maximum temperature achieved during the compression phase of the combustion cycle is somewhat decreased. By heating the combustion air prior to compression the cooling effect of the evaporation of methanol is counteracted so that greater proportions of methanol can be used. The air may be heated either before or after mixture with the methanol, and heating is preferably carried out using indirect heat exchange either with the exhaust gases or with the engine coolant liquid. Heating using indicrect heat exchange with the exhaust gases has the advantage that it is quicker to reach operating temperatures after starting and that it is self compensating with load. On the other hand the use of the engine coolant as a source of heat has the advantage that it represents a constant and stored source of heat which will continue to heat the air or air/methanol during long periods of idling.

Although heating the air before it is mixed with the methanol will produce improvements in performance, better results may be obtained if, where practical, the air/methanol mixture is heated. Evaporation of methanol lowers the temperature of the air/methanol mixture and therefore a higher inlet temperature is achieved if the air/methanol mixture, rather than the air alone, is brought to the temperature of the medium in the heat exchanger. The higher the temperature at the inlet port, the higher will be the final temperature upon compression.

The methanol can be supplied to the combustion space in any convenient way. The methanol can, if desired, be injected via a fuel injection nozzle directly into the combustion space in the cylinder or, in the case of an "indirect injection" compression-ignition engine, it may be injected into a separate chamber communicating with the space in the cylinder. Injection into the combustion space can be effected by means of a suitably timed fuel injection nozzle disposed to discharge each shot of methanol through the injection nozzle during part or all of the inlet period. Timed injection is particularly suitable in a two-stroke or a turbo-charged four-stroke engine in which a proportion of the induced air is used for scavenging, in order to avoid methanol being present in the exhaust gases. In this case the air alone may be heated by heat exchange in the inlet path with waste heat from the engine. Preferably, however, the methanol is also heated, and this may be carried out by arranging for the injection to take place onto a "hot spot" in the engine. Such a "hot spot" may be caused deliveratelv by the absence of ducts for circulation of coolant in a small area.

Another convenient method of introducing the methanol is to form the methanol/air mixture prior to its introduction into the combustion space. Thus, for example, the methanol can be metered to a further fuel nozzle disposed in the air inlet path, the methanol being fed to the fuel injection nozzle via a fuel injection pump, In this case the fuel injection nozzle can be arranged to deliver each "shot" of methanol onto a "hot spot" in the wall of the air intake heated by the exhaust gases. In this way the methanol is wholly or largely flash vapourised prior to entry into the combustion space. Alternatively the methanol can be fed to a nozzle or jet disposed in the walls of a venturi in the air inlet path to the combustion engine. Conveniently the methanol/air mixture may be produced using a carburettor from which the usual butterfly valve may be removed, if desired. Yet another alternative for introducing the methanol into the air stream is to use an ultrasonic generator delivering into the air inlet path. It will be appreciated that introducing the methanol into the induced air prior to its introduction into the combustion

space can be used in four stroke engines but can only be used in two stroke engines if provision is made for the scavenging air to be free of methanol either by providing a separate source of air for scavenging or by injecting methanol only intermittently so that the scavenging air is free of methanol.

It will be apparent that, in operating the method of the invention, the compression-ignition engine will be supplied simultaneously with methanol and diesel fuel from two separate fuel tanks or a fuel tank having two compartments, one for the methanol and one for the diesel fuel. However, since the calorific value of methanol (10,500 BTU/1b (gross) —8580 BTU/1b) is approximately one half that of diesel fuel (19,000 BTU/1b 18,330 BTU) a larger than usual tank must be fitted for the methanol in order to obtain the same endurance as compared with a similar engine operating on diesel fuel only.

The diesel fuel may be injected into the combustion space in the conventional manner via a conventional fuel injector nozzle or via two or more such nozzles, to which it is metered in the usual way by a conventional fuel-injection pump. In the case of an "indirect injection" engine, it may be ejected into the separate chamber.

We have also found that diesel engines can be run satisfactorily, using the two fuels methanol and diesel with methanol supplying more than 66% by weight of the total fuel quantity if a quantity of a cetane improver is incorporated in the diesel fuel.

According to the present invention in a second aspect there is provided a method of operating a compression-ignition engine having at least one combustion space comprising the steps of providing a mixture comprising methanol and air in the said combustion space, compressing the mixture, and injecting a quantity of a diesel fuel containing a cetane improver into the at least partially compressed mixture.

Example of cetane improver are alkyl nitrates, such as amyl nitrate, peroxides hydroperoxides, epoxides, such as propylene oxide, and hydrazine.

The amount of cetane improver to be added to the diesel fuel may be varied but is usually from about 0·1 to about 5% by volume of the diesel fuel, for example from about 0·5 to 2% by volume, more especially about 1% by volume.

The diesel fuel is injected into the at least partially compressed methanol/air mixture in the combustion space. The precise moment of injection of the diesel fuel may depend on numerous factors including the design of the engine (and, in the case of reciprocating piston type of compression-ignition engine, particularly the design of the piston head), the load of the engine and the engine speed. However, preliminary experiments indicate that in a reciprocating piston type of compression-ignition engine, injection should usually take place in the range between a crank angle of 30° and a crank angle of 5° before top-dead-centre, e.g. at 16° before top-dead-centre. The engine may be fitted with an injection advance device such as is provided by the CAV rotary distributor fuel injection pump.

The relative proportions of the diesel fuel and methanol to be used depend on the design of the engine, the load and the engine speed. The proportions of methanol and diesel fuel for a particular load and/or engine speed can radily be determined for a particular engine by a process of trial and error. Once the desired proportions have been determined at all operating speeds and loads of the engine a suitable mechanical linkage can be devised for connecting the diesel fuel- and methanol-metering devices in order to be able to achieve the desired ratio at any given speed of the engine and engine load.

The methanol fed to the engine may be essentially pure methanol or it may contain without harm proportions, e.g. up to 25% by weight of one or more minor ingredients such as water, ethanol, propanol, isopropanol and/or butanol. Thus a typical methanol-containing fuel may contain 85% by weight methanol the balance comprising water, ethanol and higher alcohols. It may also contain a proportion, e.g. up to 10% or more by weight of diesel fuel or other hydrocarbon or vegetable oil(s) and sufficient of an emulsifying agent to produce a uniform composition. The methanol may also contain a dye and/or a denaturing agent e.g. pyridine.

The engine may be of the "indirect injection" type in which a separate chamber is provided communicating with the remainder of the combustion chamber, into which separate chamber the diesel fuel is injected. However it is considered to be preferable that where practical the head should be a direct injection low swirl head in order that the heat lost to the cylinders and head walls due to turbulence may be minimised.

The engine may be fitted with a turbo-charger or a plurality of turbo-chargers connected in series, the or each turbo-charger optionally being fitted with an intercooler.

In another aspect the invention comprises a compression ignition engine comprising at least one combustion chamber provided with a fuel injection nozzle, means for metering diesel fuel to the or each fuel injection nozzle, means for introducing methanol into the or each combustion chamber and heat exchange means for increasing the temperature of air supplied to the or each combustion chamber by indirect heat exchange with waste heat from the engine. Such an engine may be connected to suitable sources of methanol and diesel fuel to form a power unit. Preferably the diesel fuel contains a cetane improve such as

amyl nitrate. The methanol may contain one or more of the additives listed above.

The invention will now be described, by way of example, with reference to the drawings accompanying the provisional specification, in which:—

Figure 1 is a graph showing the variation of smoke emission (expressed in Bosch numbers) and brake specific fuel consumption (B.S.F.C.) with brake means effective pressure (B.M.E.P.) when an experimental engine is run as a normal diesel engine;

Figure 2 is a graph showing the variation of gross fuel consumption expressed in Kilograms/hour) and of heat input (in megajoules/hour) with brake means effective pressure (B.M.E.P.) in bars;

Figure 3 shows a graph illustating the variation of percent diesel fuel (gas oil) (calculated on an energy basis) with speed to give the same maximum (B.M.E.P.) as when the engine is run as a normal diesel engine;

Figure 4 illustrates the variation of percent full load gas oil with speed to give a similar maximum (B.M.E.P.) as when the engine is run as a standard diesel engine at a Bosch 3 smoke emission level;

Figure 5 is a graph showing the variation of delay period with inlet temperature at constant load;

Figure 6 shows the effect on delay period upon adding 1% by volume of amyl nitrate to the gas oil at different air inlet temperatures;

Figure 7 shows a diagram of the general arrangement of the power unit of a Bedford 220 truck converted to enable it to operate on the dual-fuel principle according to the invention;

Figure 8 shows the heat exchanger;

Figure 9 shows a section on the line A—A of Figure 8;

Figure 10 shows a schematic diagram of the control system for the engine;

Figure 11 shows a diagram of the modification of the diesel fuel pump controls;

Figure 12 shows a diagram of the selector control as fitted to the carburettor; and

Figure 13 shows a cross-section of the carburettor.

The engine used for the tests, the results of which are shown in Figures 1 to 6, was a Ricardo E16 single cylinder 121mm × 140 mm direct injection engine having a cylinder swept volume of 1595 cm³ and a compression ratio of 15·3 to 1. The combustion chamber was a toroidal deep bowl type having a diameter of 2·5 times the depth. The injector used for the diesel fuel was a standard 4-hole nozzle centrally placed. The engine had two valves for the single cylinder controlled by push rods and operated on the 4-stroke cycle. A Zenith (Registered Trade Mark) 42W downdraught carburettor was used with a fixed wide open throttle supplying the methanol fuel to the engine air inlet. The quantity of methanol was manually controlled by a threaded tapered needle valve.

In experimental investigations such as the present a certain difficulty arises from the use of two separate fuels which have different densities and calorific values. Either all fuel quantities must be expressed in heat values or, for practical convenience, the quantity of the second fuel can be expressed as an equivalent quantity, having the same heat quantity of the first fuel.

For the present tests, as the performance comparisons are with a standard diesel engine fuelled exclusively with "gas oil" distillate, quantities of methanol used have been expressed as the equivalent quantity, calorifically, of gas oil.

Figure 1 shows a typical load range curve taken at 36·3 rev/s on the experimental engine used operating as a normal diesel engine using only gas oil for fuel. Plotted on the same figure are the smoke levels of the exhaust as determined in Bosch units. The higher the Bosch number the greater the concentration of soot in the exhaust gases.

The same basic information is replotted in Figure 2 in a different form. Here the gross fuel consumption is shown plotted against the brake mean effective pressure. It is found that the plot of gross fuel quantity used, or alternatively, the heat content of the fuel used is a straight line at the lower loads (bmep's) and only deviates at the higher loads where the combustion efficiency deteriorates as a result of the decreasing amount of oxygen available. This is shown by the presence of a rapidly increasing soot concentration in the exhaust gases at high loads. It is found that if the straight line is projected backwards below zero bmep it intercepts the abscissa axis at a negative bmep value which is fairly closely equal to the engine's motoring less at the particular speed at which the load range curve has been taken. The fuel consumed at zero bmep is that value required to just overcome all the engine's friction losses.

For convenience the value of bmep at which the exhaust soot concentration was Bosch 3, which is a typical maximum value permitted in service, was taken to represent full load when operating on gas oil alone. The quantity of gas oil corresponding to this condition may be called 100%.

At other test conditions when diesel fuel was injected as the ignition source with methanol aspirated from the carburettor the quantity of diesel fuel used was fixed for each test. For convenience the fixed quantity of gas oil used has been expressed as × % (usually 40%) of the quantity needed for full load at the same engine speed when operating entirely as a normal diesel engine. Load variation during a given test was obtained by varying only the quantity of methanol in-

duced by means of a variable jet in the carburettor body.

Initial tests (Figure 3) showed that the quantity of methanol, as a percentage of the total fuel used, required to give a performance similar to that when using diesel fuel only was quite high, about 75% of the total fuel, at low engine speed but decreased rapidly so that at maximum engine speed only about 30% of methanol could be used.

A detailed study indicated that the inability to burn larger proportions of methanol at the higher speeds was a rapid increase in the ignition delay as the proportion of methanol increased. Simple changes in injection timing have relatively little effect in overcoming this problem.

Measurements showed the compression pressures to be 36·3 bars for full diesel operation and 32·4 bar when methanol was induced. Measurement of temperature within the inlet valve head port showed this to be 316°K as a pure diesel and 283°K with induced methanol. The lower value with methanol is due to the high latent heat of evaporating metahanol. Deriving compression indices from the compression pressure and using these values to estimate the compression temperatures at value of 770°K is found for pure diesel operation and only 610°K with methanol being induced.

In view of the much reduced compression temperatures, a trial was made of the effect of air inlet temperature on the ability to reduce delay and run on larger proportions of methanol. Figure 4 shows that the amount of gas oil required to give full load output at the maximum experimental speed of 36·7 rev/s diminished as the inlet temperature is raised, the fuel oil injection timing being kept fixed. At the top temperature of 90°C tried experimentally it was found that the injection timing could be advanced by 7° to give a further reduction in the percentage of fuel oil required to obtain full load. But at these higher inlet temperatures small variations in mixture strength or timings led to "knock" so it is not practical to operate here.

In view of these findings another series of tests were carried out to determine the actual ignition delay at various intake temperatures from observations of the period between the start of injector needle lift, i.e. start of fuel injection, and the point at which the cylinder pressure diagrams started to rise above the normal compression pressure diagram using electronic indicating equipment. These observations are shown in Figure 5. It should be noted that the inlet temperatures used as abscissae in this figure are values for the mixture temperature after the methanol has been carburetted. As might be expected, increased inlet temperatures do reduce the ignition delay but even at 70°C the delay period with 40% gas oil/methanol fuelling proportions is still significantly longer than that for 100% gas oil, as used in normal diesel engines, at ambient air temperatures. The tests at 23·3 rev/s, although not so complete show the same trend.

Further tests have shown that with heating of the carburetted methanol/air mixture it is possible to achieve satisfactory running of a compression-ignition engine using 85% methanol/15% diesel fuel (on an energy basis) at low speeds and 60% methanol/40% diesel fuel (also on the energy basis) at high speeds.

It is known that certain chemical substances such as, by way of illustration, amyl nitrate can be added in small percentages to diesel fuel to improve cold starting and smoother operation by reducing the combustion delay period. It is however not usual to use these additives commercially in normal diesel engine operation.

With this background knowledge it was decided to carry out tests with amyl nitrate added to the diesel fuel pilot charge used for compression ignition.

The curves shown in Figure 6 for 36·7 rev/s indicate that, when using a gas oil to which 1% by volume amyl nitrate has been added, the ignition delay period at 40% gas oil is reduced to a value which is slightly lower than that existing when 100% gas oil is on its own. With the quantity of methanol required to give roughly 3 bar bmep at ambient inlet temperature satisfactory operation was obtained up to 70°C inlet mixture temperature, which was the maximum temperature dried during these tests.

In view of the satisfactory reduction in ignition delay with the use of 1% by volume amyl nitrate added to the diesel engine fuel used, the quantity of methanol induced was increased with the mixture temperature in the inlet valve elbow at 22°C. It was found that ignition delay lengthened with the increased quantity of methanol but satisfactory operation occurred and an output of 6·0 bar obtained. However, upon raising the mixture temperature using the same fuel quantities it was found that the phenomenon of "knock" became observable.

Further tests have shown that satisfactory running at high speeds of a compression-ignition engine can be achieved using carburetted methanol and an addition of amyl nitrate to the diesel fuel with proportions of methanol as high as 70% on an energy basis.

We will now describe how a Bedford 220 truck can be converted to enable it to operate on the dual-fuel principle.

Referring to Figure 7, the diesel engine 11 of the Bedford truck is provided with a conventional diesel fuel pump 12 connected to the main tank 13 which contains a supply of diesel fuel.

An auxiliary tank 14 containing commercial grade methanol is fitted to the chassis of

the truck and connected to an electrically-operated S.U. (Registered Trade Mark) fuel pump 15. The S.U. fuel pump 15 supplies methanol to a modified S.U. carburettor 16 fitted in the air intake 17 of the engine 11.

As can best be seen from Figures 8 and 9 a heat exchanger jacket 18 is fitted around the air intake 17 downstream of the carburettor and connected to the exhaust manifold 19 so that the exhaust gases from the engine pass through the jacket 18 around the air intake 17, before they enter the exhaust pipe 20. The exhaust gases heat the air/methanol mixture in the air intake 17 causing the methanol to vaporise and thus providing to the engine through the inlet manifold 21 an even distribution of methanol.

As can be seen from Figure 10 the accelerator pedal 22 in the driver's cab controls by means of a Bowden cable 23 the setting of a butterfly valve 24 in a Venturi 25 in air intake 17. In this respect the arrangement is the standard arrangement on a Bedford 220 truck. The butterfly valve 24 is capable of limited movement between a fully open position and a partially closed position. The settings of the diesel fuel pump 12 and the carburettor 16 are controlled by a penumatic control system operating from the pressure drop in the Venturi 25 in the air intake 17. The low-pressure side of the Venturi 25 is connected through a pipe 26 to a chamber 27 with a spring-loaded diaphragm wall 28 (see Figure 11). The high-pressure side of the Venturi is connected through a pipe (not shown) to the chamber on the other side of the diaphragm wall so that the position of the diaphragm wall 28 varies according to the pressure drop in the Veturi. The diaphragm wall is coupled to the control rack 29 of the diesel fuel pump 12 which controls the amount of diesel fuel delivered by the injectors so that changes in the pressure drop cause corresponding movements of the rack 29 and corresponding variations in the amount of diesel fuel injected. The low pressure side of the Venturi 25 is also connected through a pipe 30 which branches from the pipe 26 to a chamber 31 with a spring-loaded diaphragm wall 32 (see Figure 13). The high-pressure side of the Venturi is connected through another pipe (not shown) to the other side of the diaphragm wall. The diaphragm wall 32 is coupled to the plunger 33 of the carburettor 16 which raises and lowers the needle valve 34 in the jet 35.

When operating a dual fuel the needle valve 34 of the carburettor 16 and the rack 29 of the diesel fuel pump 12 move at the same time and at the same rate so that by shaping the needle valve appropriately the proportion of methanol to diesel fuel can be different at different throttle settings.

Figures 10, 11 and 12 show the arrangement for switching from diesel to dual fuel operation. A selector lever 36 mounted in the driver's cab operates two Bowden cables 37 and 38 connected respectively to the diesel fuel pump 12 and the carburettor 16.

As can be seen in Figure 11 the cable 37 is connected to one end of a lever 39 which is pivoted at a point intermediate its ends. The other end of the lever 39 acts as a movable stop to limit the movement of the rack 29 of the diesel fuel pump 12. A spring 40 operating between the end of the sheath of the Bowden cable 37 and the lever biases the lever to rotate in a clockwise direction as viewed in Figure 11 so that when the selector lever 36 is moved forward into the diesel operation position, the cable 37 allows the lever 39 to abut an adjustable maximum fuelling stop 41. In this position the lever 39 allows movements of the rack up to the position 42 corresponding to the maximum desired setting for full load diesel operation.

Operation of the selector lever 36 into the dual fuel position causes the cable 37 to pull the lever 39 against bias of the spring 40 so that it rotates anticlockwise into a position in which it abuts an adjustable dual fuel stop 43. In this position the lower end of the lever 39 stops the movement of the rack 29 at the position 44 corresponding to the maximum desired setting fur dual fuel operation. The rack 29 can only then move between the position 45 corresponding to the idling setting and the position 44. It will be appreciated that the control system for switching to dual fuel operation does not interfere with the usual system for moving the rack to the position 60 in which no fuel is supplied to the engine, when it is desired to stop the engine.

The other Bowden cable 38 acts on one end of a lever 45 pivoted about a point intermediate its ends and biased by a spring 46 operating between the end of the sheath of the Bowden cable 38 and the lever 45 to rotate in an anticlockwise direction as seen in Figure 12. The other end of the lever 45 acts on the plunger 33 of the carburettor 16. When the selector lever 36 is in the diesel position, the cable 38 is moved forward and the spring 46 rotates the lever in an anticlockwise direction causing the lever 45 to depress the plunger 28 and close the carburettor jet 35. When the lever 36 is moved into the dual-fuel position the cable 38 rotates the lever 45 in the clockwise direction until it releases the plunger 33 and abuts a stop 47.

The needle valve 34 of the carburettor 16 is connected to the plunger 33 by a lost-motion coupling 48 comprising a hollow cylinder 49 fixed to the plunger and a rod 50 secured to the needle valve 34 with an enlarged end which is a sliding fit in the cylinder. The diameter of the bore of the cylinder is reduced at its lower end to form a stop which co-operates with the enlarged end of the rod 50 to return the end of the rod in the cylinder.

A spring 51 in the cylinder 49 acts between the lower end of the plunger and the enlarged end of the rod 50 to bias the needle valve downwardly towards the carburettor jet 35.

When the plunger is pressed fully down, as when the selector level is in the diesel operating position or when there is no pressure drop across the Venturi, the needle valve is firmly seated in the carburettor jet and the spring 51 is partially compressed. When the plunger 33 rises from the lowermost position the compression of the spring is reduced as the rod plunger and cylinder move up but the rod remains stationary and the needle valve remains closed. Only when the plunger has risen to the point where the reduced diameter part of the cylinder abuts the enlarged head of the rod 50 does the plunger lift the needle valve and open the carburettor jet.

The converted Beford truck operates as follows:

When operating on diesel only the carburettor is held closed so that no methanol can be used and the diesel fuel pump can operate in the usual manner delivering diesel up to the usual maximum amount permitted for diesel operation.

When operating on diesel fuel and methanol, the plunger 33 of the carburettor is allowed to rise. On start up, and idling the pressure drop in the Venturi is insufficient to take up the lost motion in the lost motion coupling 48. The carburettor jet remains closed and therefore the engine is fuelled by diesel alone operating in the same was as if the engine had not been converted for dual fuelling. As the throttle is opened further or the load is increased the carburettor valve opens and the carburettor needle and the diesel fuel pump rack move at the same rate, but the relative proportions of diesel to methanol are different at different settings owing to the shaping of the carburettor needle. For any particular pressure drop the amount of diesel delivered will be the same as for operation on diesel alone, but the power delivered by the engine will be greater because of the introduction of methanol. On a further increase in the pressure drop across the Venturi the rack of the diesel fuel pump will abut the movable stop formed by the lever 39 preventing the amount of diesel injected from being increased beyond the maximum for dual fuel operation (say about 50% of the maximum for diesel only operation). Further increase in the amount of methanol introduced into the air intake is permitted on increase in the pressure drop in the Venturi until a shoulder 52 on the plunger 33 in the carburettor abuts an adjustable stop 53.

WHAT WE CLAIM IS:—

1. A method of operating a compression-ignition engine having at least one combustion space which method comprises providing a mixture comprising methanol and air in the said combustion space, compressing the mixture and injecting a quantity of a diesel fuel into the at least partially compressed mixture, the temperature of the air being raised by at least 20°C prior to compression, by preheating with waste heat from the engine.

2. A method according to claim 1 in which the temperature of the air is raised by at least 50°C.

3. A method according to claim 2 in which the temperature of the air ir raised by at least 90°C.

4. A method according to claim 3 in which the air is preheated by the engine coolant liquid.

5. A method according to any one of the preceding claims in which the said preheating is carried out after mixture of the methanol with the air.

6. A method according to any one of the preceding claims in which the diesel fuel includes a quantity of a cetane improver.

7. A method according to claim 6 in which the cetane improver includes one or more of an alkyl nitrate, a peroxide, a hydroperoxide, an epoxide and hydrazine.

8. A method according to claim 7 in which the cetane improver is amyl nitrate.

9. A method according to any one of claims 6 to 8 in which the amount of cetane improver is between 0·1 and 5% by volume of the diesel fuel.

10. A method according to any one of the preceding claims, wherein the said mixture is provided by carburettors of the methanol with the air.

11. A compression ignition engine comprising at least one combustion chamber provided with a fuel injection nozzle, means for metering diesel fuel to the or each fuel injection nozzle, means for introducing methanol into the or each combustion chamber and heat exchange means for increasing the temperture of air for the or each combustion chamber by indirect heat exchange with waste heat from the engine.

12. An engine according to claim 11 in which the heat exchanger is connected to the engine coolant system.

13. An engine according to claim 11 or claim 12, wherein the heat exchanger is arranged so as to heat the said air by heat exchange with the engine exhaust gases.

14. A power unit comprising an engine according to any one of claims 11 to 13 connected to a source of diesel fuel and a source of methanol.

15. A power unit according to claim 14 in which the diesel fuel contains a cetane improver.

16. A power unit according to claim 15 in which the cetane improver is amyl nitrate.

17. A power unit according to claim 15 or 16 in which the amount of cetane improver is from 0·1 to 5% by volume of the diesel fuel.

18. A method of operating a compression ignition engine substantially as described hereinbefore with reference to the drawings accompanying the provisional specification.

19. A power unit substantially as described hereinbefore with reference to the drawings accompanying the provisional specification.

BOULT, WADE & TENNANT,
27 Furnival Street,
London, EC1A 1PQ.

Printed for Her Majesty's Stationery Office by Burgess & Son (Abingdon), Ltd.—1981.
Published at The Patent Office, 25 Southampton Buildings, London, WC2A 1AY,
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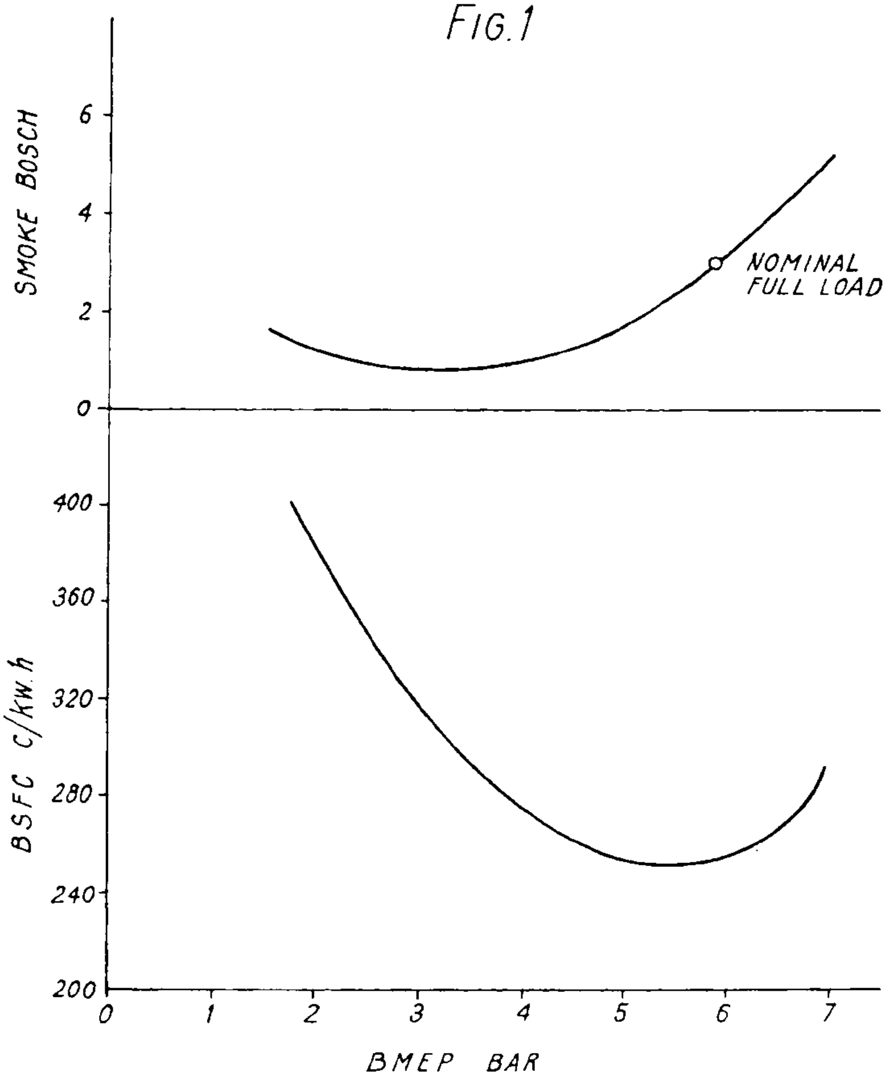
FIG. 1


SMOKE BOSCH
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2
0
NOMINAL
FULL LOAD
400
360
320
280
240
200
BSFC c/kw.h
0
1
2
3
4
5
6
7
BMEP BAR

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Sheet 2

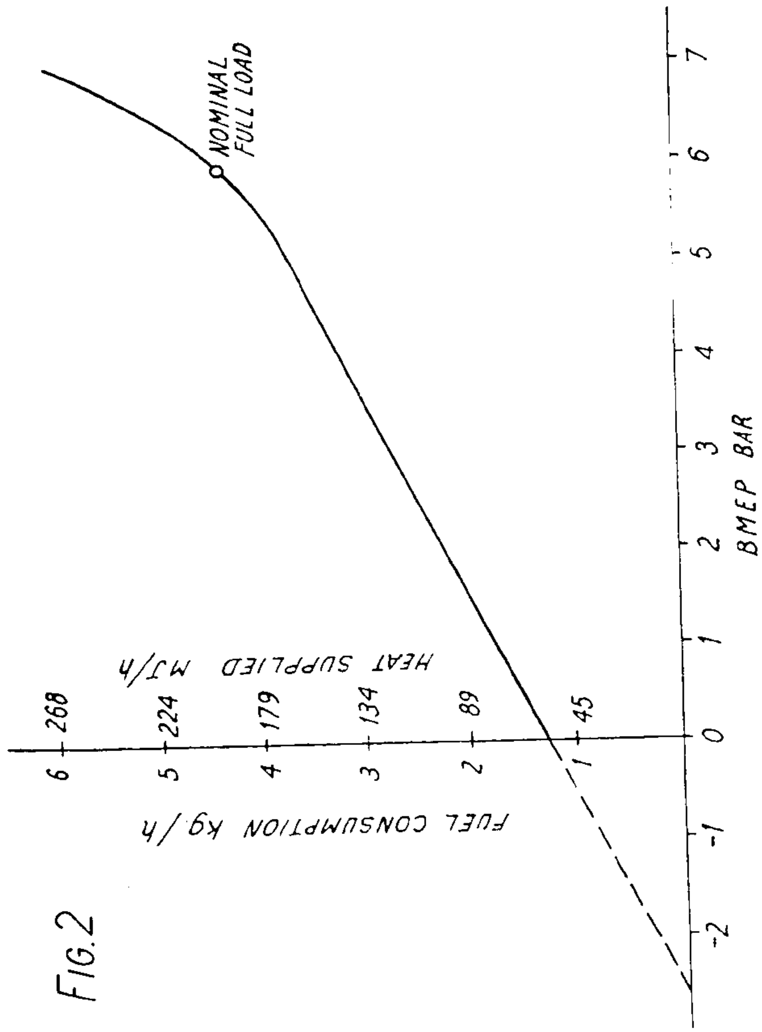


FIG.2

FIG.3

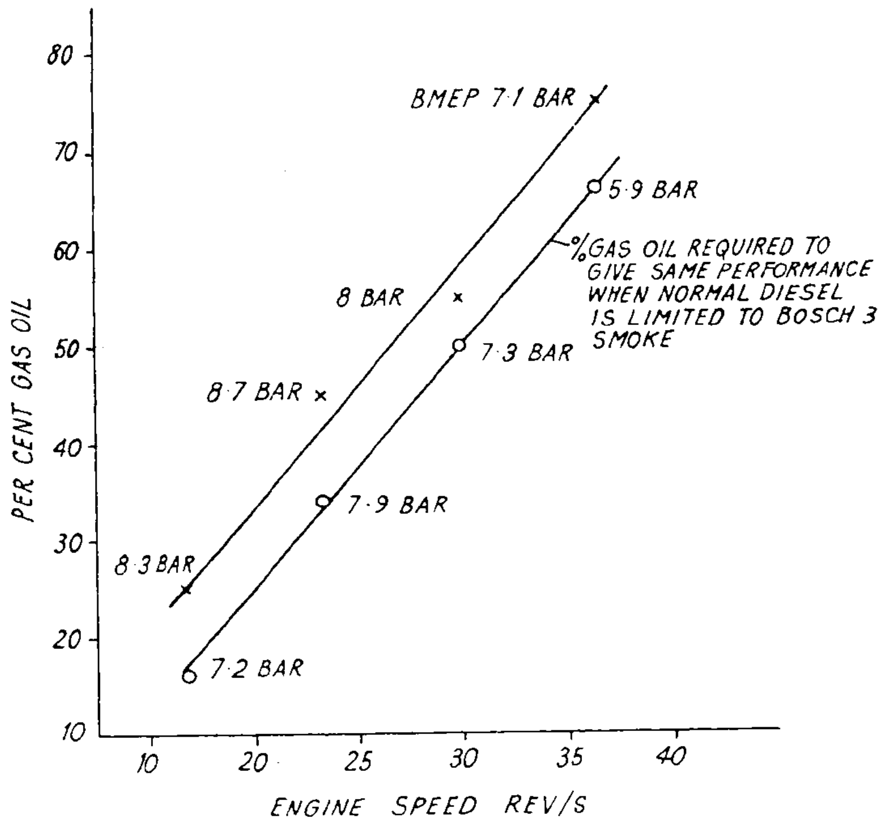

BMEP 7·1 BAR
5·9 BAR
%GAS OIL REQUIRED TO GIVE SAME PERFORMANCE WHEN NORMAL DIESEL IS LIMITED TO BOSCH 3 SMOKE
8 BAR
7·3 BAR
8·7 BAR
7·9 BAR
8·3 BAR
7·2 BAR
PER CENT GAS OIL
80
70
60
50
40
30
20
10
10
20
25
30
35
40
ENGINE SPEED REV/S

FIG. 4

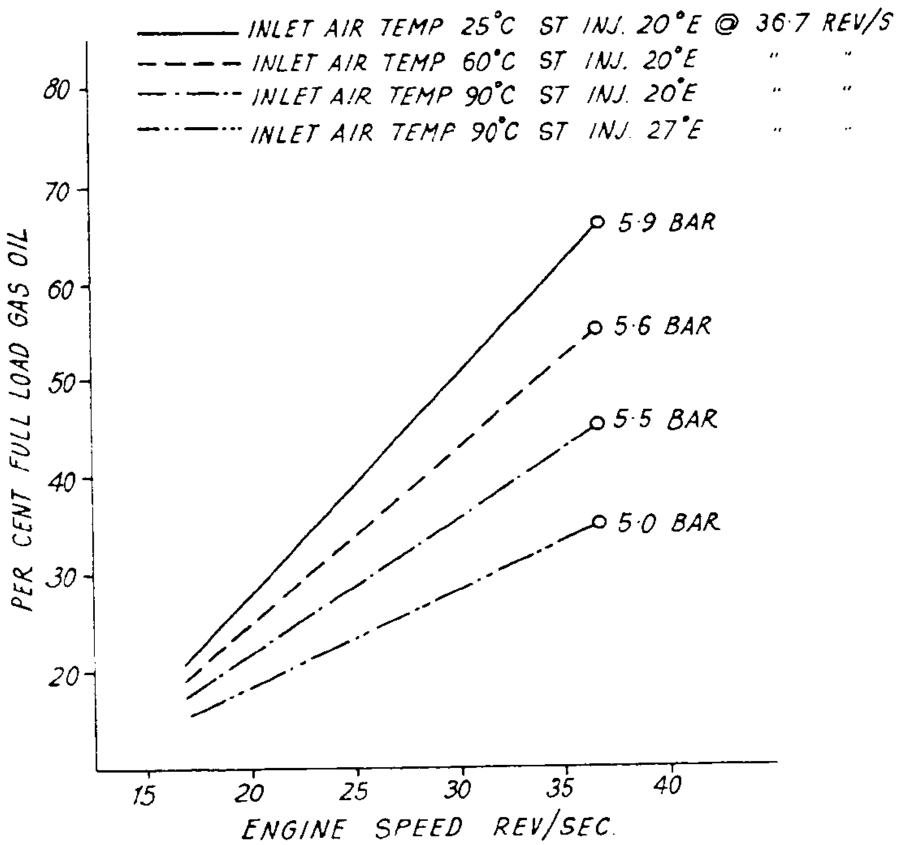

INLET AIR TEMP 25°C ST INJ. 20°E @ 36·7 REV/S
INLET AIR TEMP 60°C ST INJ. 20°E " "
INLET AIR TEMP 90°C ST INJ. 20°E " "
INLET AIR TEMP 90°C ST INJ. 27°E " "
5·9 BAR
5·6 BAR
5·5 BAR
5·0 BAR
PER CENT FULL LOAD GAS OIL
80
70
60
50
40
30
20
15
20
25
30
35
40
ENGINE SPEED REV/SEC.

FIG.5

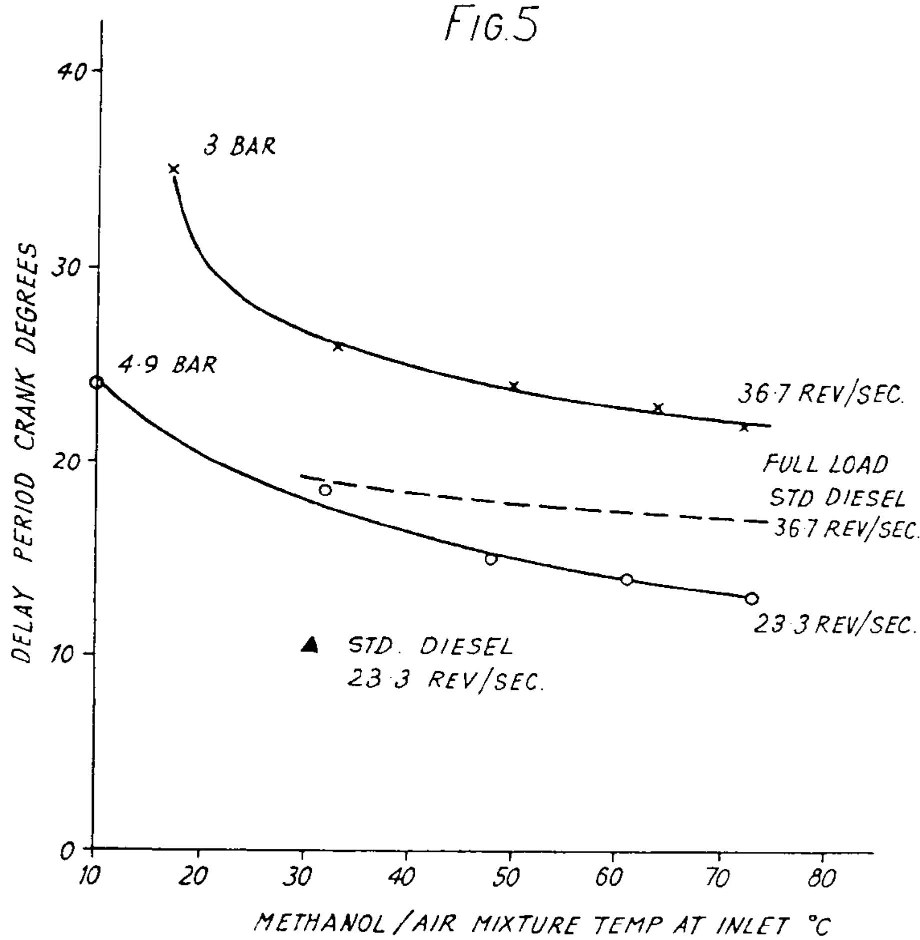

40
30
20
10
0
DELAY PERIOD CRANK DEGREES
3 BAR
4·9 BAR
36·7 REV/SEC.
FULL LOAD
STD DIESEL
36·7 REV/SEC.
23·3 REV/SEC.
STD. DIESEL
23·3 REV/SEC.
10
20
30
40
50
60
70
80
METHANOL/AIR MIXTURE TEMP AT INLET °C

FIG. 6

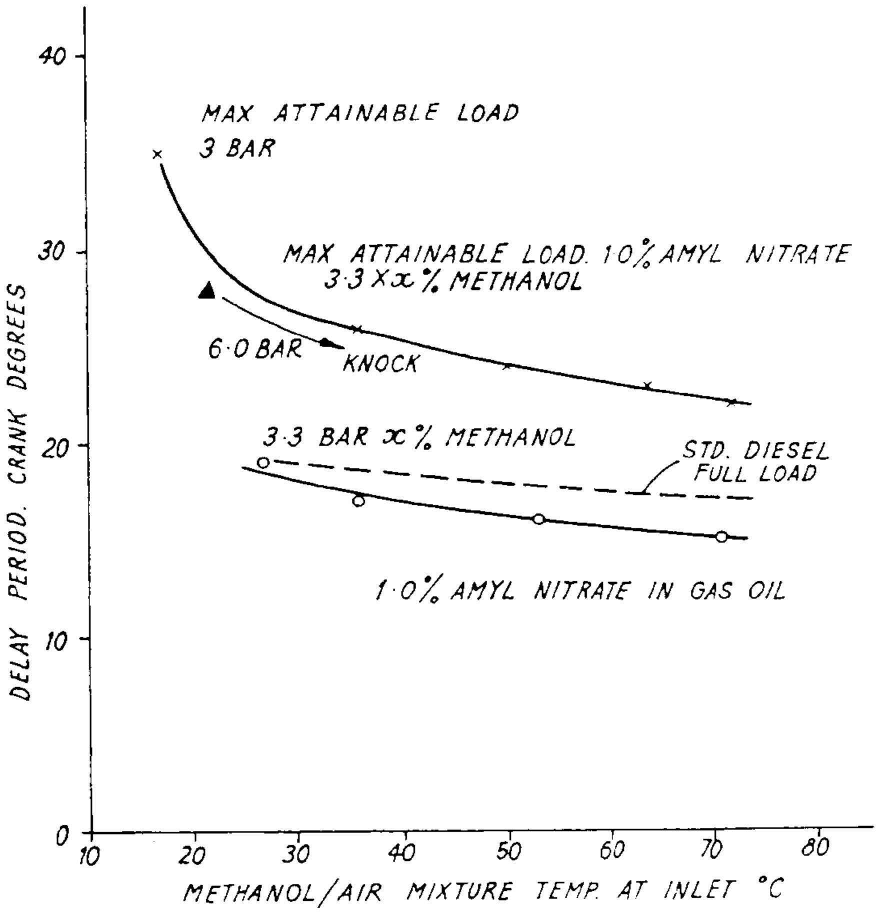

MAX ATTAINABLE LOAD
3 BAR
MAX ATTAINABLE LOAD. 1.0% AMYL NITRATE
3.3 X x % METHANOL
6.0 BAR
KNOCK
3.3 BAR x % METHANOL
STD. DIESEL
FULL LOAD
1.0% AMYL NITRATE IN GAS OIL
DELAY PERIOD. CRANK DEGREES
METHANOL/AIR MIXTURE TEMP. AT INLET °C
0
10
20
30
40
50
60
70
80

FIG.7

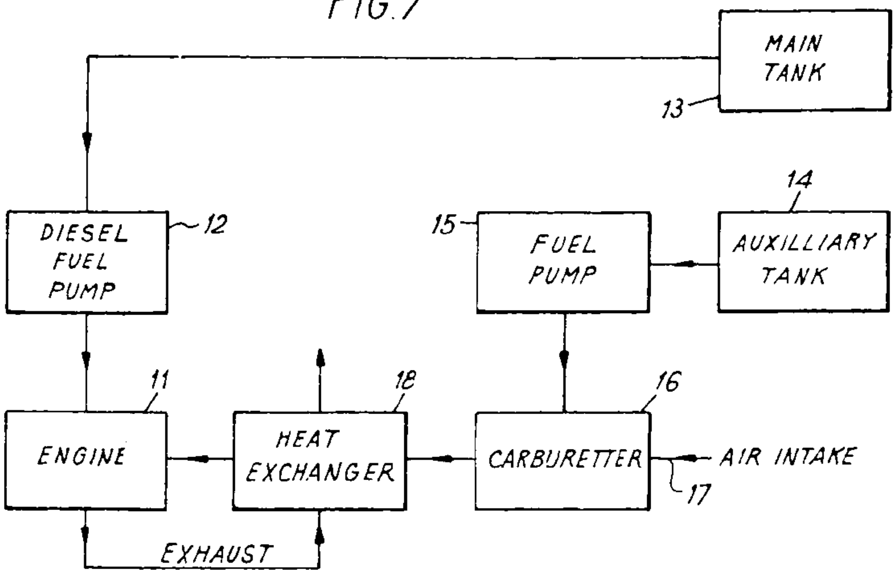


FIG.8

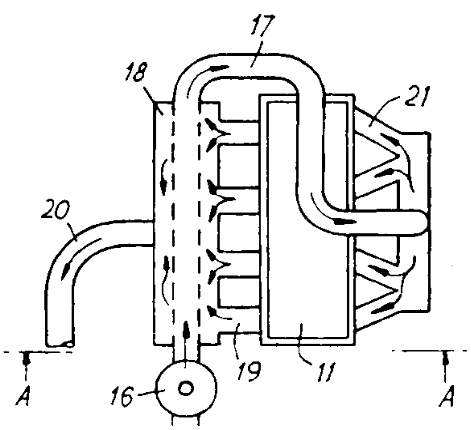


FIG.9

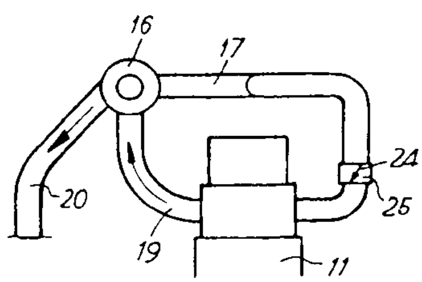


FIG. 10

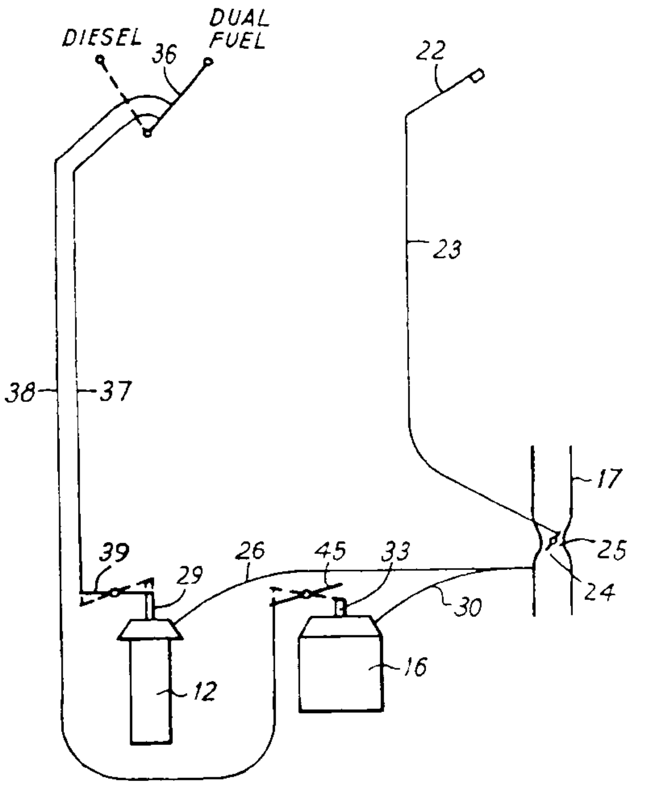


FIG. 11

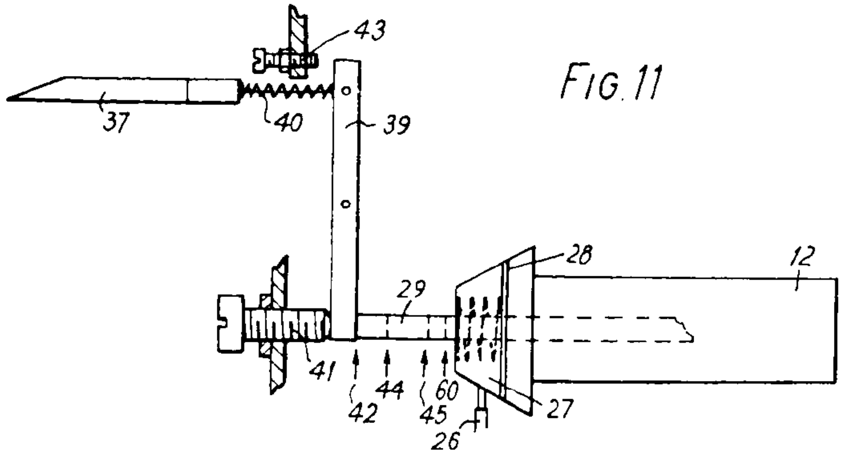
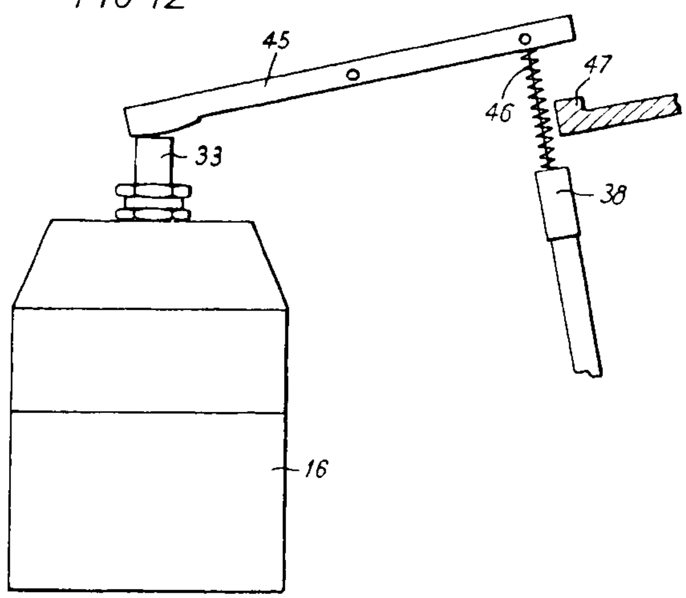


FIG 12



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Sheet 10

FIG. 13